AD-A116 785 COLUMBIA UNIV NEW YORK LUBRICATION LAB F/G 11/8
AN INVESTIGATION OF THE FUNDAMENTAL PROPERTIES OF GREASE-LUBRIC--ETC(U)
MAY 82 H G ELROD, R H BUCKHOLZ N00014-79-C-0407

UNCLASSIFIED NL

1 OF 1
AD A116785

END
DATE FILMED
8-82
DTIC

AD A116785

NR. 0??-013

(13)

COLUMBIA UNIVERSITY
DEPARTMENT OF MECHANICAL ENGINEERING
LUBRICATION LABORATORY

AN INVESTIGATION OF THE FUNDAMENTAL PROPERTIES OF GREASE-LUBRICATED BEARINGS

U. S. OFFICE OF NAVAL RESEARCH
Contract N00014-79-C-0407

FINAL REPORT
May 1982

H. G. Elrod, Princ. Investigator

R. H. Buckholz, Sr. Investigator

PREFACE

H. G. Elrod

Funds for the present contract were intended as "seed money" for a more extended investigation into the properties of grease-lubricated bearings, and into the possibilities of the use of grease with compliant bearing surfaces. The need for such information is great. A substantial fraction of all bearings used in the U. S. Navy employ grease as lubricant. Despite this fact, the design of these bearings is much more uncertain than that for conventional fluid-film bearings operating with oil. Research in grease has been deemed less attractive because of the experimental and analytical difficulties involved. Reproducible application of lubricating grease is more difficult by virtue of its stiffness, and analysis is more difficult because of its non-Newtonian, thixotropic character. So the job proposed for this contract remains to be done. Grease Lubrication needs from the ONR the same sort of support ONR gave Air Bearings. In that case, ONR nurtured a collaborative research program involving governmental and industrial research which took Air Bearing design from art to science.

The present report describes preliminary results obtained by modifying and reactivating an apparatus originally built by Smith and Fuller for investigating turbulent lubrication. All the work described herein was carried out by Prof. R. H. Buckholz as his initial experimental research in our Dept. of Mech. Engrg. He has since obtained financial support from the National Science Foundation to investigate the performance of oil in compliant-surface journal bearings. For this purpose, the apparatus has been further modified and more fully instrumented. We hope later to pursue again the possibilities of grease in conjunction with compliant surfaces, since we believe this combination offers great promise. In addition, unsupported analytical research is currently being conducted on non-Newtonian bearing behavior.

PRELIMINARY EXPERIMENTAL STUDY OF GREASE-LUBRICATED JOURNAL BEARINGS

R. H. Buckholz

SUMMARY:

Grease is used as a lubricant for large numbers of bearings in everyday use; but the non-Newtonian rheological properties of these thixotropic gels, and their effect upon journal-bearing performance, are not well understood. To initiate a program in grease lubrication, we reactivated a concentric-cylinder journal-bearing test apparatus used by Smith and Fuller. This particular configuration has been treated extensively by theoreticians for constant viscosity lubricants, and to a considerably lesser extent for non-Newtonian lubricants. Smaller amounts of experimental data exist for both types of lubricants.

The journal-bearing test apparatus was operated first as a concentric-cylinder viscometer, and the shear viscosity of SAE 10 oil measured. Subsequently, the apparent shear viscosity of a litium-based grease (Sears and Roebuck) was measured as a function of rate of strain and of temperature. Performance of the grease-lubricated journal bearing was characterized by experimental determination of the coefficient of friction as a function of the bearing number ZN/p. As expected, the high shear viscosity of the grease (with corresponding high bearing number) was found to permit operation at lower RPMs.

Our preliminary results do not represent new findings. Rather, they confirm anticipated modes of operation, and suggest possibilities for improved experimental design and instrumentation for our later work.

INTRODUCTION:

The lubricating effect of grease is not easily characterized, primarily because the fluid mechanics of these thixotropic, non-Newtonian gels are too complex for the classical continuum approach, except in the simplest cases. Hence, as a start we chose a "simple case" consisting of a journal bearing test apparatus with two circular cylinders, one inside the other, in relative motion. The small gap between the two cylinders is filled with a lubricant. In the present

preliminary report, we are concerned with the general effects of grease compared with oil.

THE APPARATUS AND METHODS:

To facilitate the carrying out of a preliminary experimental program, an existing journal-bearing test apparatus (originated by Smith and Fuller) was modified to use SAE 10 oil as a lubricant. A view of the journal-bearing test section is shown in Fig. 1. The journal diameter is 2.0050", and the angular rotation can be varied from 400 to 1200 RPM. The brass bearing is machined to a diameter of 2.0095". The bearing is constrained from rotation by an external force applied 6.4" from the bearing center. The internal flow can be considered to be the superposition of a point vortex, a rigid body rotation, and a Poiseuille flow. The relative importance of each of these flows is determined by the position of the journal and the bearing, by the velocity of the journal, and by the lubricant viscosity.

The force transducer shown in Fig. 1 detects the total force, F, required to prevent the bearing housing from rotating. A typical calibration curve for this force transducer is shown in Fig. 2. A hydrostatic oil film transmits a vertical load to the bearing housing. The resultant hydrostatic film leaves the bearing housing free to rotate so that $F\ell$ exactly balances the torque caused by the fluid shear force acting on the bearing surface. The resulting balance provides a measure of the mean friction force acting on the bearing.

In all cases, except the concentric case, the fluid shear force on the bearing is a function of azimuthal position. The average friction force, f, acting on the bearing, is obtained from:

$$f = F\ell/r$$

where $$f = \iint \tau_{r\theta}\, dA$$

and $\tau_{r\theta}$ is the tangential shear stress on the bearing surface.

The journal bearing is instrumented with ten thermocouples, as shown in Fig. 3. Azimuthal and axial temperature distributions within the bearing surface can then be sequentially measured. These temperature measurements serve as a guide for estimating oil viscosity and apparent grease viscosity, but most unequivocally are not the

true oil or grease temperatures.

Measurements of shear viscosity for SAE 10 oil were made under zero-load conditions when the apparatus can be considered as concentric circular cylinders.

A pressurized air line was then used to pump grease continuously into the cylinder gap. In these experiments, the Sears Lithium-Based Grease No. 2 was used. The test apparatus was again operated as a viscometer and measurements of the apparent shear viscosity were made at temperatures between 110 deg. F and 121 deg. F. at shear rates between 18,000/s and 46,000/s. The bearing housing was then loaded and journal bearing performance experiments were conducted. Coefficient of friction data were measured as a function of bearing number.

RESULTS:

For the SAE 10 oil, no material properties were available from the manufacturer. Consequently, oil viscosity as a function of temperature for various shear rates was measured using a Rotovisko Viscometer. The results are shown in Fig. 4. Subsequent operation of the apparatus as a bearing yielded the friction coefficients presented in Fig. 5. The values and trends are consistent with expectations, as shown in Fig. 6, taken from ref. 1.

The test apparatus was next operated as a viscometer to determine the apparent shear viscosity of a lithium-base grease procured from Sears and Roebuck. Temperature and rate of strain were the independent variables. The method used for calculating the shear viscosity is given in the Appendix. Results are shown in Fig. 7, and are in qualitative agreement with those obtained by others. See, for example, Rippel (1976), from which Fig. 8 is taken.

In a final set of experiments, the grease-lubricated bearing characteristics were measured. Data for the bearing friction coefficient as a function of bearing number are presented in Fig. 9. The ranges of the variables extend from $c_f = 0.02$ to $c_f = 0.03$, and from $ZN/p = 2 \times 10^3$ to $ZN/p = 3 \times 10^3$.

DISCUSSION

When operated with a Newtonian lubricant - SAE 10 oil - the journal bearing exhibited the usual friction coefficient vs bearing parameter behavior shown in Fig. 5. When used as a concentric-cylinder viscometer to measure oil shear viscosity, the apparatus yields results consistent with those obtained on a Rotovisko.

In contrast, for the range measured, grease behaves as a non-Newtonian fluid. The decrease of shear viscosity with increasing rate of strain is exemplied by the variation shown in the curves in Fig. 8, taken from ref. 2. These results are in qualitative agreement with other investigators.

Our most significant data concern the variation of friction coefficient with bearing number. The scatter in these data can be associated with the azimuthal variation of grease viscosity as influenced by the azimuthal variation of rate of strain and of temperature. Exact measurements of azimuthal shear viscosity effects and the influence of the pressure distribution could not be obtained with the instrumentation available.

For estimation of the shear viscosity required for the abscissa in Fig. 9 (bearing number), the average rate of strain in the bearing, and the temperature from a single "reference" thermocouple, were used. Perhaps the scatter would be reduced by the introduction of a dimensionless parameter, in addition to the bearing parameter, which would reflect further the departure of the lubricant from Newtonian behavior. One such parameter might be the rate of change of the shear viscosity with rate of strain, normalized by the average viscosity divided by the average rate of strain.

For a complete set of experimental data peculiar to grease-lubricated journal bearings, a significant improvement of the test apparatus is suggested. The first improvement would be to provide for continuous measurement of the grease film thickness and of the azimuthal pressure distributions.

$$T = \frac{4\mu R_1{}^2 R_2{}^2 \Omega \pi \ell}{R_1{}^2 - R_2{}^2}$$

The small-gap approximation is:

$$T = 2\mu R_1{}^2 R_2 \Omega \pi \ell / d$$

where $d = \Delta R$.

Thus:

$$\mu = Td/2\pi R_1{}^2 R_2 \ell \Omega$$

REFERENCES

1. Fuller, Dudley D., "Theory and Practice of Lubrication for Engineers", (1956), John Wiley and sons, New York.

2. Rippel, Harry C., "Some Insights and Guidelines in the Performance Capability of Grease-Lubricated Bearings", (1976), Report for Franklin Institute Laboratories for Research, Philadelphia, Penna.

NOTATION

c_f - Friction coefficient (f/total upward force)

f - Total shear force on bearing surface
F - Force measured by transducer at arm-length, ℓ force)
N - Journal rotation rate, RPM
P - Unit bearing pressure (lbf/in^2), being upward force/projected bearing area.
Z - Lubricant shear viscosity

APPENDIX

During sustained operation, a hydrodynamic shear force acts on the surface of the bearing. The bearing housing is supported on a "friction free" hydrostatic shoe. The fluid shear force will act to rotate the bearing housing; this effect is balanced by a force F applied at a distance ℓ from the center of the bearing (See Fig. 1). The average shear force, f is determined from the following equation

$$f = F\ell/r$$

The flow between two 4-inch concentric cylinders of radius R_1 = 1.0025 in. and R_2 = 1.005 in. was established using the bearing test apparatus. The torque on the outer cylinder is related to the bearing dimensions R_1, R_2, the torque arm, ℓ, the lubricant viscosity, μ, and the operating speed, Ω, by:

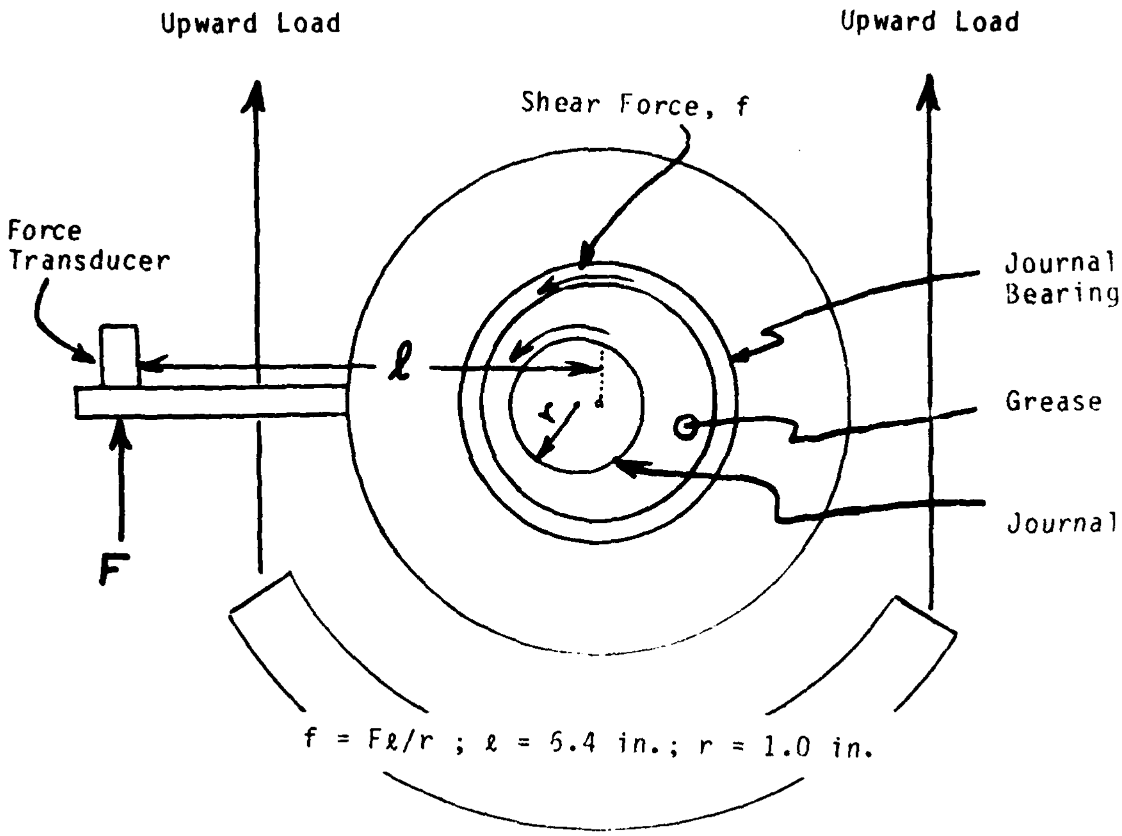


FIG. 1 SIDE VIEW OF JOURNAL-BEARING TEST APPARATUS
(length scales are not mutually consistent)

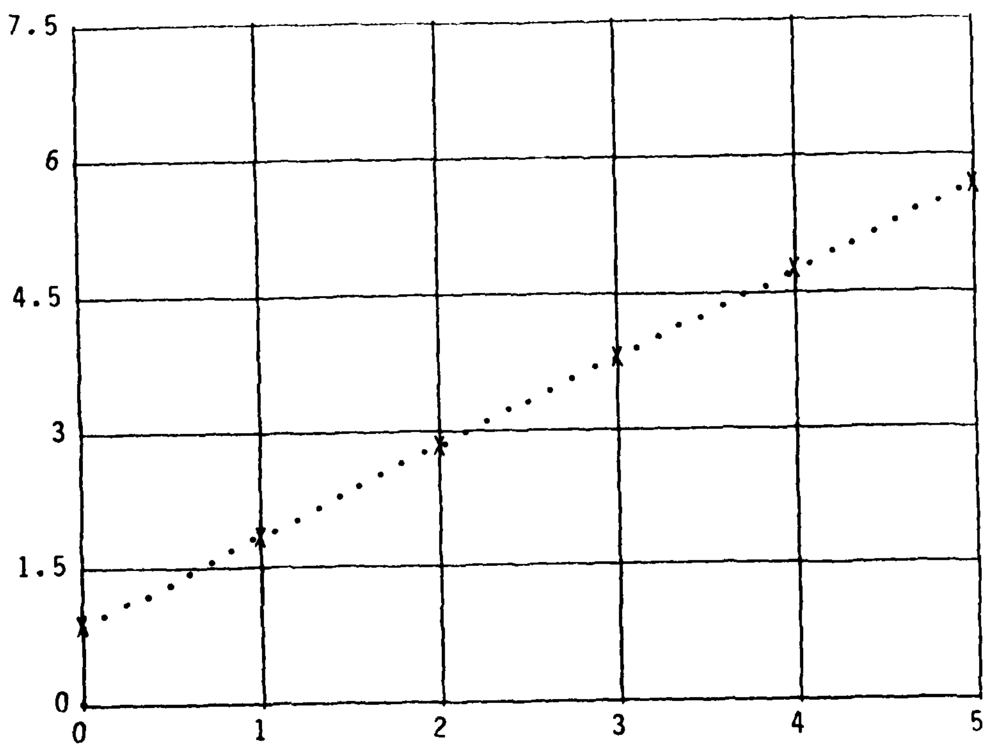


FIG. 2 TRANSDUCER MILLIVOLTS*E-1 VS. LOADING (LBF)*E 0
TYPICAL CALIBRATION CURVE FOR THE FORCE TRANSDUCER
SHOWN IN FIG. 1

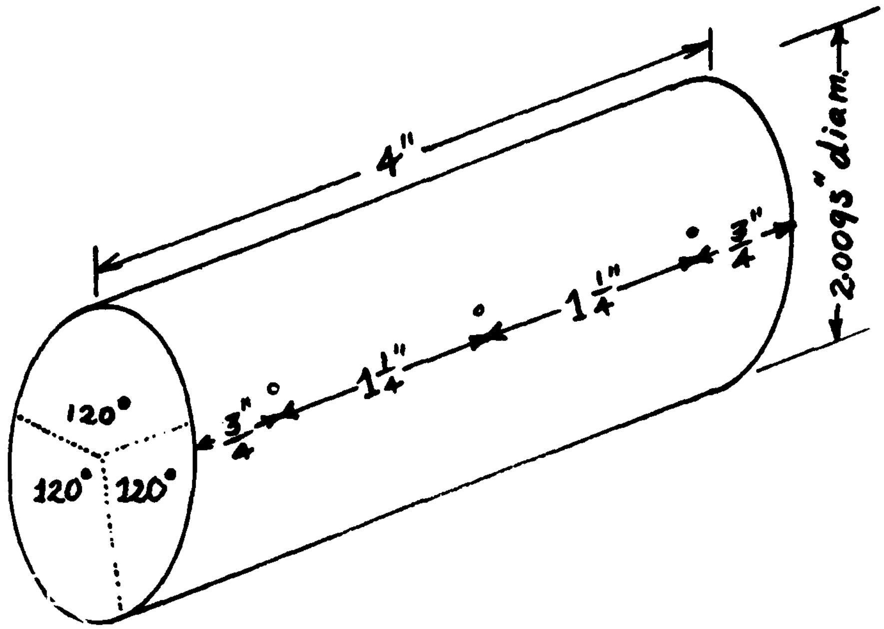


FIG. 3 LOCATION OF BEARING THERMOCOUPLES

(Three rows of thermocouples were placed 1/16 in. from the lubricated surface. A tenth couple was placed near the outer radius of the bearing to estimate temperature distribution. Length scales not mutually consistent.)

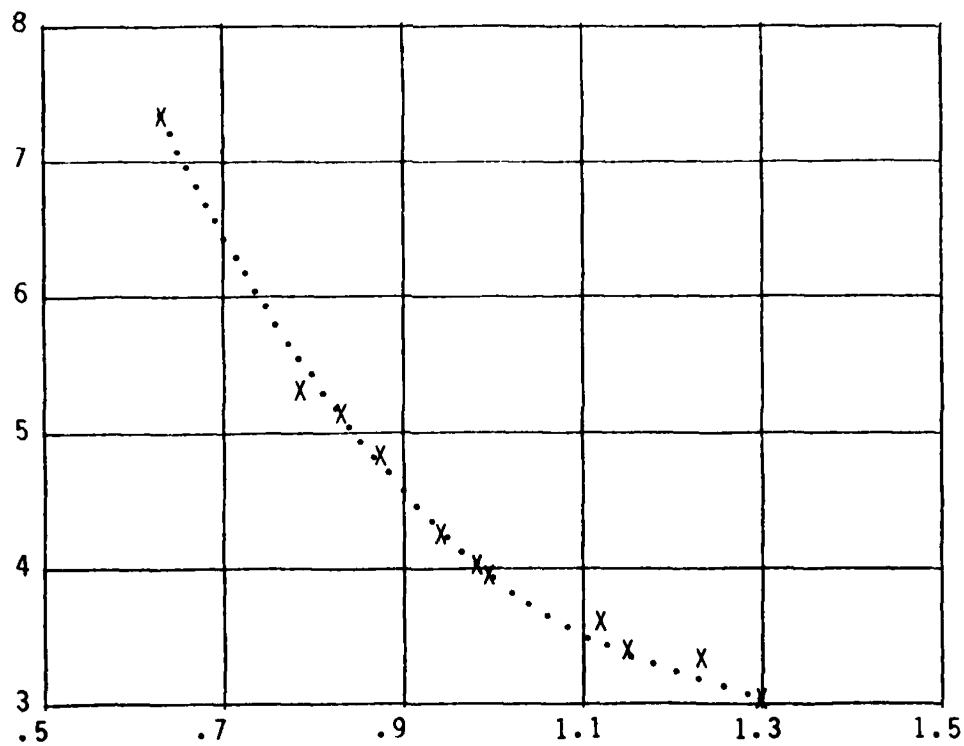


FIG. 4 OIL VISCOSITY (CP)*E-1 VS. TEMP. F.*E-2
AT A FIXED STRAINRATE OF 685 SEC^{-1}

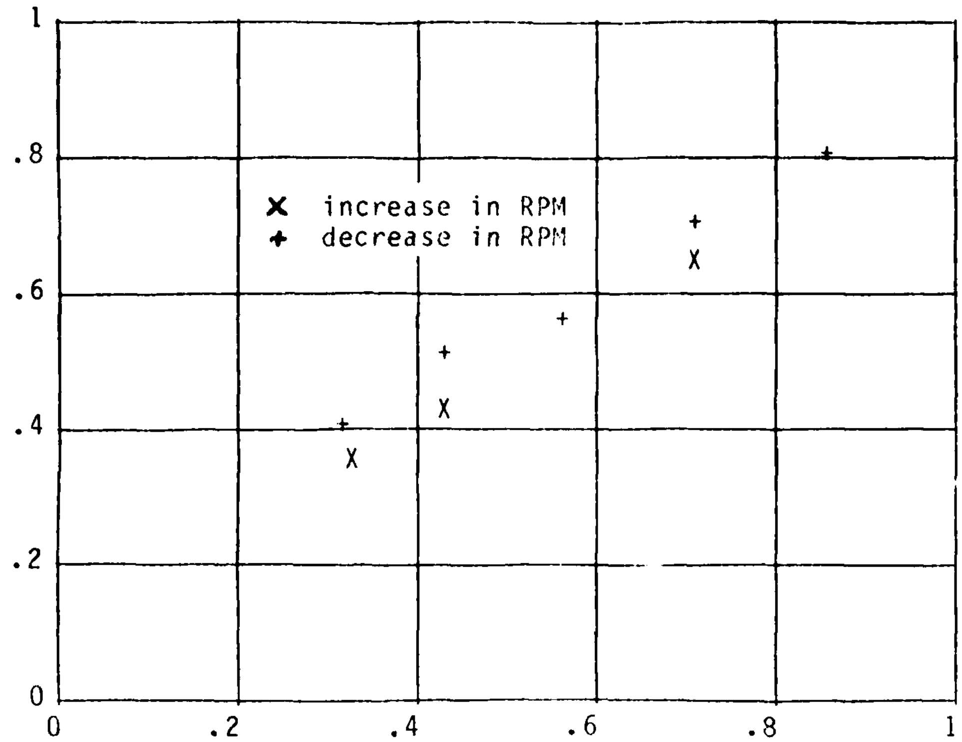


FIG. 5 COEFF. FRICTION*E 3 VS. BRG. NO. (ZN/P)*E-3
COEFFICIENT OF FRICTION FOR AN SAE 10 OIL
AS FUNCTION OF BEARING NUMBER

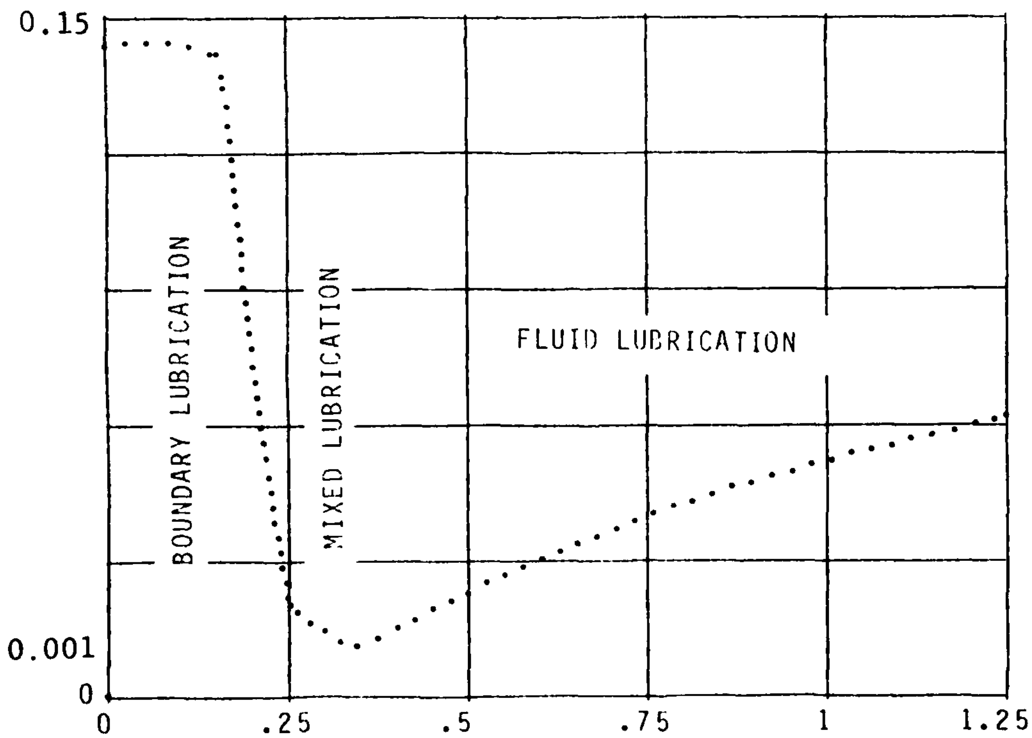


FIG. 6 COEFF. FRICTION VS. BEARING NO.*E-2
COEFFICIENT OF FRICTION AS FUNCTION OF BEARING NO.
FOR OIL-LUBRICATED JOURNAL BEARINGS (ref. 1)

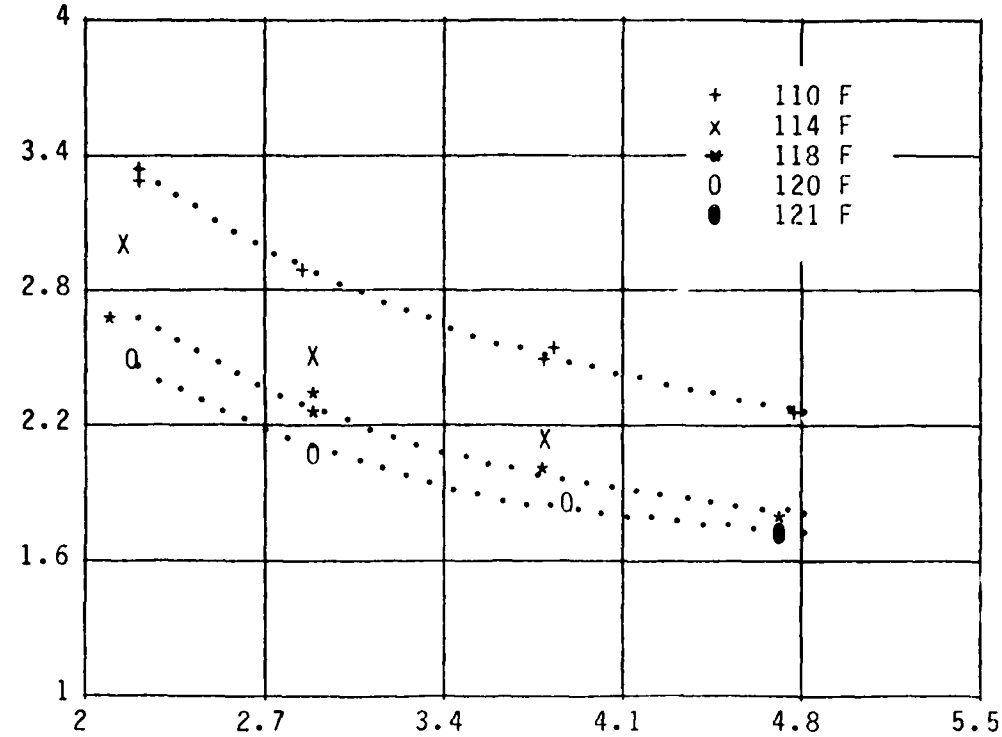


FIG. 7 GREASE VISC.*E-2 VS. STRAINRATE*E-1

APPARENT GREASE VISCOSITY (cp) AS FUNCTION OF STRAINRATE (sec^{-1}) AND TEMPERATURE, F.

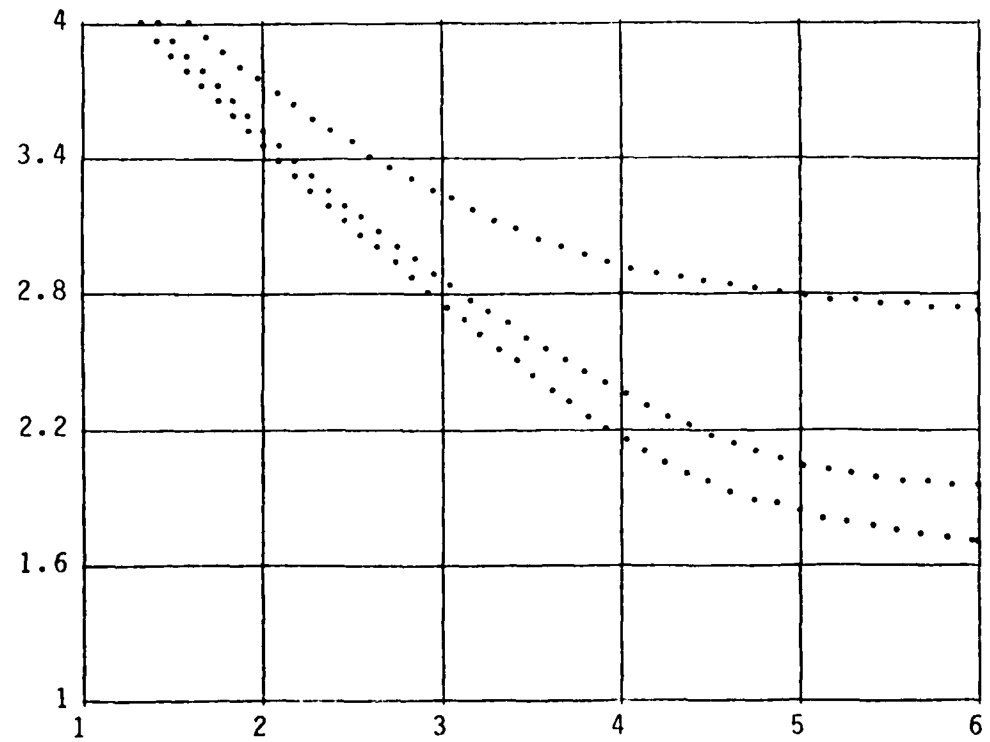


FIG. 8 LOG(APPARENT VISC.)*E 0 VS. LOG(STRAINRATE)*E 0

APPARENT VISCOSITY OF SUN PRESTIGE 7-H EP GREASE
CONSISTING OF OIL + SPERM + LEAD NAPTHANATE
APPROXIMATELY 6% SOAP, 94% FLUIDS

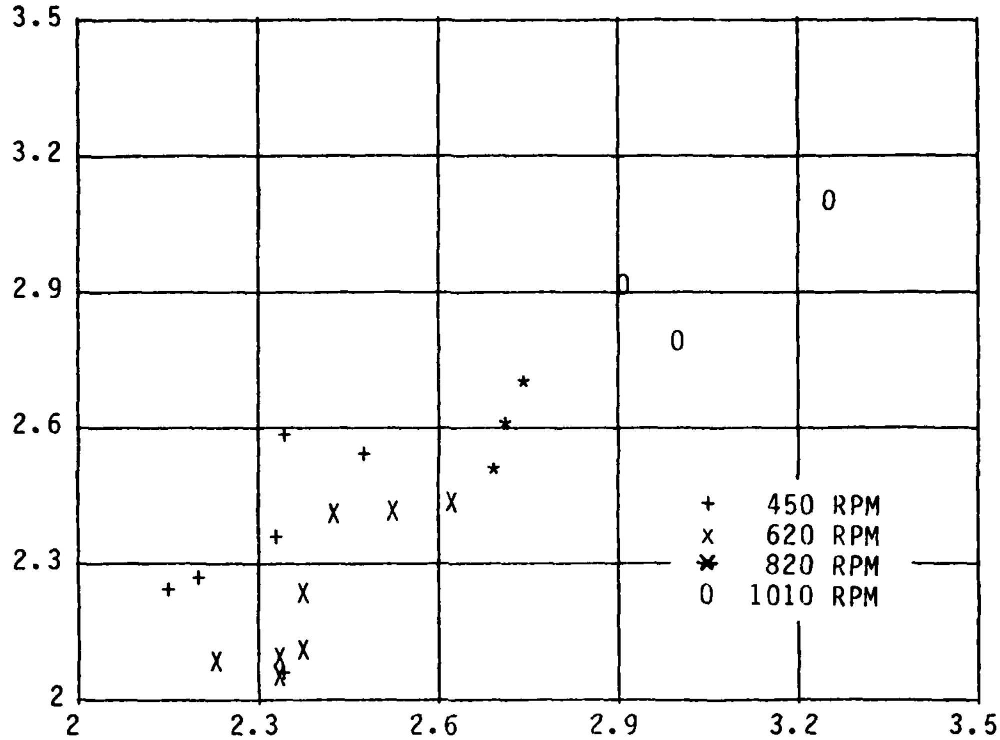
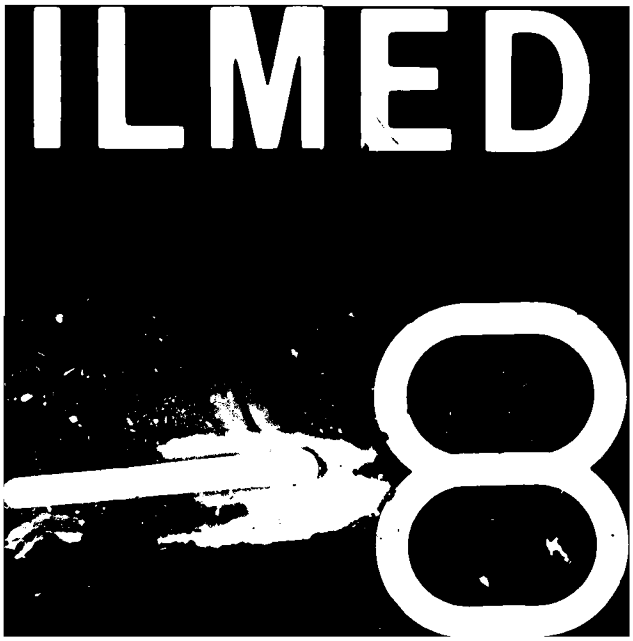


FIG. 9 COEFF. FRICTION*E 2 VS. BRG. NO. (ZN/P)*E-3
DATA REDUCED FOR FOUR ROTATION SPEEDS


ILMED
8